AD/A-003 273

DEVELOPMENT OF EXPLOSIVE RIPPER WITH TWO-STAGE COMBUSTION

Ronald J. Mathis

Southwest Research Institute

Prepared for:

Army Mobility Equipment Research and Development Center

October 1974

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER AD/A 003273
4. TITLE (and Subtitle) DEVELOPMENT OF EXPLOSIVE RIPPER WITH TWO-STAGE COMBUSTION		5. TYPE OF REPORT & PERIOD COVERED Final Report- May - Oct. 74
		6. PERFORMING ORG. REPORT NUMBER AR 965
7. AUTHOR(s) Ronald J. Mathis		8. CONTRACT OR GRANT NUMBER(s) DAAK-02-74-0263
9. PERFORMING ORGANIZATION NAME AND ADDRESS Southwest Research Institute P. O. Drawer 28510 San Antonio, TX 78284		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 11-3952-001
11. CONTROLLING OFFICE NAME AND ADDRESS U.S. Army Mobility Equipment Research and Development Center Fort Belvoir, Virginia 22060		12. REPORT DATE October 1974
		13. NUMBER OF PAGES 45 + preliminarys & Appendix
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

This document has been approved for public release and sale; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

same as 16

18. SUPPLEMENTARY NOTES

None

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Rock ripping
Two-Stage Combustion
Ripper
Spoils Removal

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report summarizes the development of an explosive ripper with two-stage combustion. This effort was conducted for the U. S. Army Mobility Equipment Research and Development Center under Contract DAAK 02-74-0263

DD FORM 1473 EDITION OF 1 NOV 65 IS OBSOLETE

i

DEVELOPMENT OF EXPLOSIVE
RIPPER WITH TWO-STAGE COMBUSTION

by

R. J. Mathis

FINAL REPORT

D D C

October 1974

7 1975

Approved:

ellWood

C. D. Wood, Director
Department of Engine and
Vehicle Research
Automotive Research Division

iii

ABSTRACT

This report summarizes the development of an explosive ripper with two-stage combustion. This effort was conducted for the U. S. Army Mobility Equipment Research and Development Center under Contract DAAK 02-74-0263.

ib

TABLE OF CONTENTS

Table of Contents (cont'd.)

ii

LIST OF ILLUSTRATIONS

SYMBOLS AND DEFINITIONS

SYMBOLS

BFF	-	Back Flow Function
e	-	Natural Logarithm
k	-	Ratio of Specific Heats
M	-	Mass
p	-	Sage Pressure
P_b	-	Ratio of Combustion Pressure to Charge Pressure
P_c	-	Charge Pressure
T	-	Temperature
V_1	-	First Stage Volume
V_2	-	Second Stage Volume
V_3	-	Third Stage Volume
x	-	Distance Along Combustion Chamber
x_0	-	Total Distance of Combustion Chamber
α	-	Ratio of Final Stage Volume to Total Volume
ΔM	-	Change of Mass
ΔT	-	Change of Temperature
Δx	-	Change of Position
ρ	-	Density

I. INTRODUCTION

This report explains the work accomplished under Contract DAAK02-74-0263, "Development of Explosive Ripper with Two-Stage Combustion," from 1 May 1974 to 31 October 1974. The staged combustion technique was conceived at Southwest Research Institute, and is an extension of an earlier concept of increasing the rock-ripping capability of the standard D7F Caterpillar tractor.

The objective of this contract was two-fold; one, to demonstrate that the entire fuel/air explosion system could be conveniently mounted on the D7 ripper tractor. Secondly, to produce the same combustion pressure as obtained on previous REDSOD (Repetitive Explosive Device for SOil Displacement) devices using a small 20 psig blower and a diesel fuel/air mixture. Previously designed systems using the explosive fuel/air concept incorporated a large 1200 CFM, 100 psig blower with a propane/air mixture. This blower was towed alongside the tractor, drastically reducing its maneuverability while increasing the operating cost.

The operating cycle for the stage combustion device is illustrated in Figure 1. The entire system is charged to 20 psig air pressure by the tractor driven blower. Upon the operator's command the diesel fuel is sprayed into the combustion chambers. The mixture is then ignited at one end of the large combustion chamber, as the mixture burns and expands along the tube, the unburned mixture is compressed and forced into the smaller chamber. As the flame front progresses, a strong ignition source is generated in the smaller chamber by flame passing through the interstage check valve. This flame rapidly ignites the mixture in the smaller chamber which has been compressed to nearly 100 psig, resulting in a final combustion pressure of approximately 600 psig. This combustion pressure drives open the exhaust valve, allowing the combustion gas in the chamber to be released through the duct work into the rock formation being fractured. The system is then purged and the cycle can be repated on a single shot basis or in an automatic firing mode, by the operator's command. The small 20 psig, 350 scfm blower provides an automatic firing rate of one shot every 6 seconds.

The project results will be presented in three basic sections of this report. First, the analysis and model testing of the staged combustion concept. Next, the design and equipment development necessary to adapt the device to the tractor. Finally, a discussion of results of the staged combustion device concept.

Charging Chambers

CHARGING AIR (20 PSIG)

INLET CHECK VALVE

EXHAUST VALVE

FUEL SPRAY (200 PSIG)

2nd STAGE CHAMBER

IGNITION SOURCE

FLAME PROPAGATION

1st STAGE PEAK COMBUSTION PRESSURE 85-100 PSIG

1st. Stage Combustion

2nd STAGE PEAK COMBUSTION PRESSURE 550-800 PSIG

INTERSTAGE CHECK VALVE

2nd Stage Combustion and Gas Release
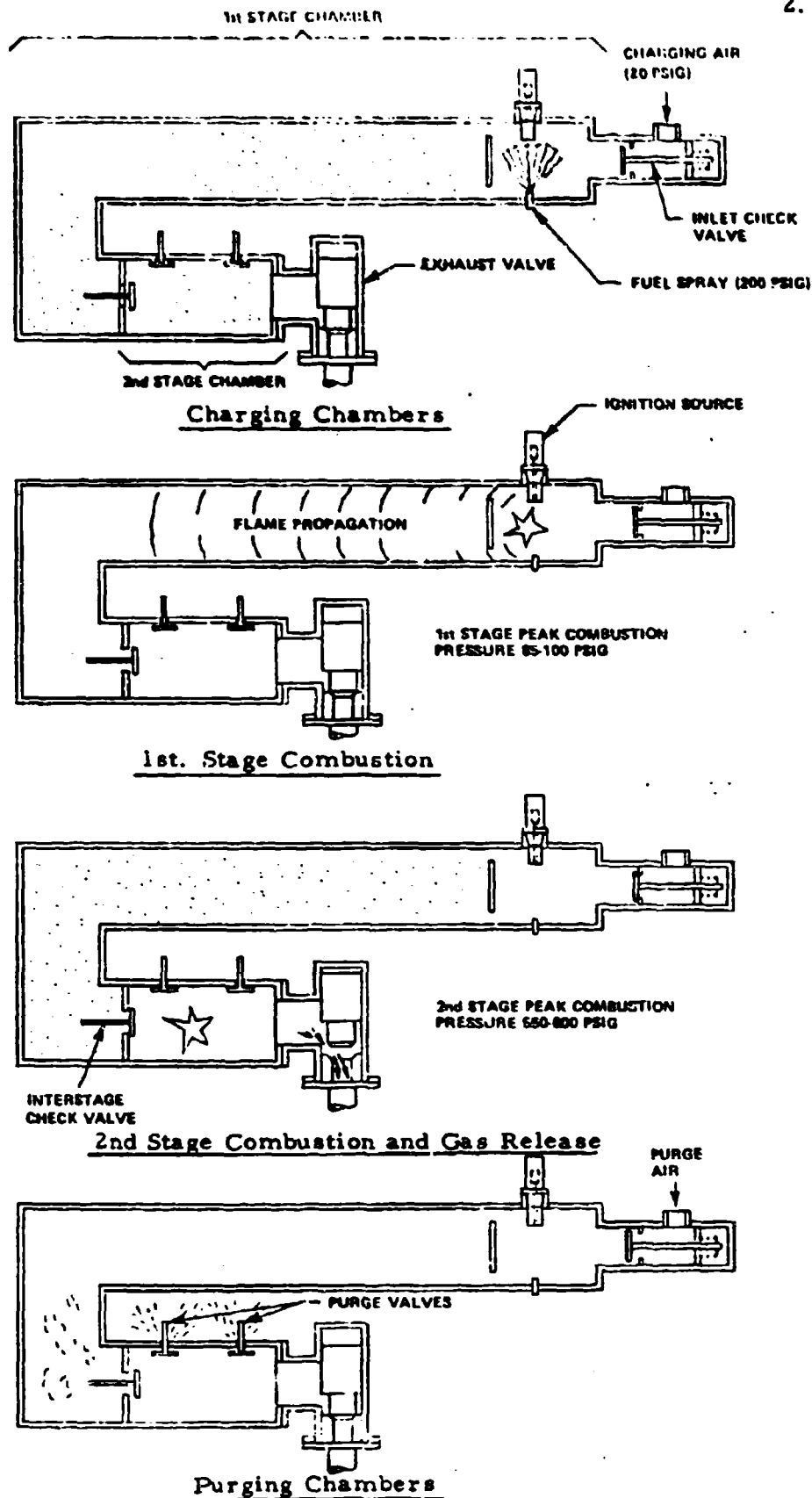
PURGE AIR

PURGE VALVES

Purging Chambers

FIGURE 1. TWO-STAGE COMBUSTION OPERATING SEQUENCE

II. ANALYSIS AND MODEL TEST

A.   Analysis

       The stage-combustion concept is based on the premise that
two or more combustion chambers are connected together by an orifice,
and both are filled with a combustible mixture of fuel and air. An
ignition source located in one end of the larger chamber ignites the
mixture. The flame front progresses away from the igniter and as
it burns the pressure in the chamber increases. The increased
pressure causes the unburned mixture to flow from the first chamber
to the succeeding chambers. When the flame front reaches the orifice,
the pressure of the unburned gas in the succeeding chambers will be
higher than its original press ire. The flame then passes through
the orifice and ignites the mixture. The final pressure in the end
volume, after combustion, is dependent upon the volume ratio of
the combustion chambers and the initial charging pressure of the
system.

       ` order to establish the performance level of the stage-
combust `. concept, a theoretical analysis was performed for two
different operating conditions. The first condition analyzed consisted
of the combustion chambers connected together by an orifice; this
condition assumes that back-flow of the combustion gases occurs and
the final pressure obtained is the same in each of the chambers.
The second condition analyzed assumes an infinite number of check
valves in each chamber, eliminating the back-flow of gas. The com-
bustion pressure would therefore continue to increase for each stage
of combustion. This type of operation would eventually be realized
by the use of a check valve between each stage of combustion.

       The detailed analysis of these two operating conditions are given
in Appendix A with the final equations shown below:

$$P_b = \rho^k \left(1 + \frac{k\Delta T}{\rho^{k-1}}\right) \qquad \text{(pressure-density relation)}$$

$$\alpha = -\frac{1}{\frac{1}{\rho e^{(k-1)\Delta T}}(\rho^{k-1} - 1)} \qquad \text{(no back flow condition)}$$

$$\alpha = \frac{1}{\rho}\left(1 + \frac{1}{k\Delta T}\right) - \frac{1}{k\Delta T}\rho^{k-1} \qquad \text{(back flow condition)}$$

These equations represent the maximum combustion pressure that may be obtained for each stated condition. Both equations are shown in Figure 2 as the ratio of the pressure rise versus the combustion volume ratio. It is apparent from Figure 2 that for a combustion volume ratio between 0.5 and 1.0, there would be no advantage to the use of interstage check valves. However, for volume ratios less than 0.5, the elimination of the back flow becomes significant. These equations define the upper limits of the stage combustion process for a quiescent mixture and do not account for the heat losses to the surrounding chamber walls.

B.   Model Test

1.   Description

The analysis presented in the previous section indicated that the stage-combustion process is capable of producing combustion pressures of 600 psia for a charge pressure of 15 psig, assuming a volume ratio of 0.2. The most significant question that arises from this study is as follows: Will an actual process produce this high a final pressure using the same charge pressure and volume ratio?

To answer this question, models similar to the one shown in Figure 3 were constructed. These models consisted of various sections of pipe fittings used for combustion chambers, connected by either an orifice, or a floating valve. Each chamber was instrumented with a separate pressure transducer to determine peak combustion pressures and the pressure rise rate.

Initially, the chambers were charged to a predetermined level using a propane-air mixture. The mixture was then ignited using a standard automotive type spark plug. This ignition system also served as a trigger for the oscilloscope. A typical pressure trace for a two-stage system connected by an orifice is shown in Figure 4. Figure 5 shows the same test configuration with the exception of an interstage check valve located between the two chambers. This procedure was repeated using both two and three stage systems, connected by an orifice or a check valve while varying the charge pressure and volume ratio. The pressure levels obtained using these models are shown in Table I for the two-stage system connected by an orifice, and Table II for the two and three-stage systems using a check valve between the chambers.
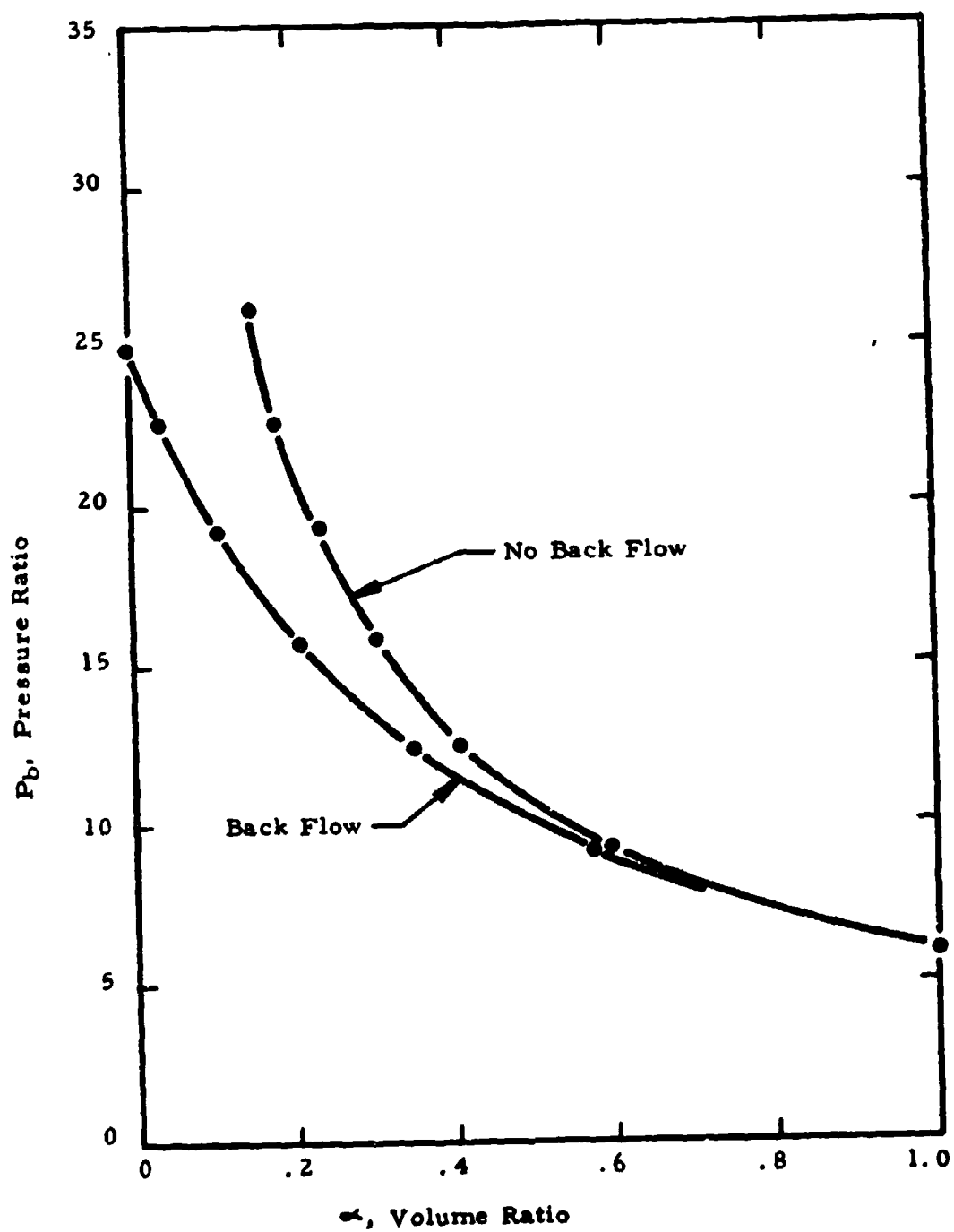
FIGURE 2. THEORETICAL CURVE OF PRESSURE RATIO VS. VOLUME RATIO
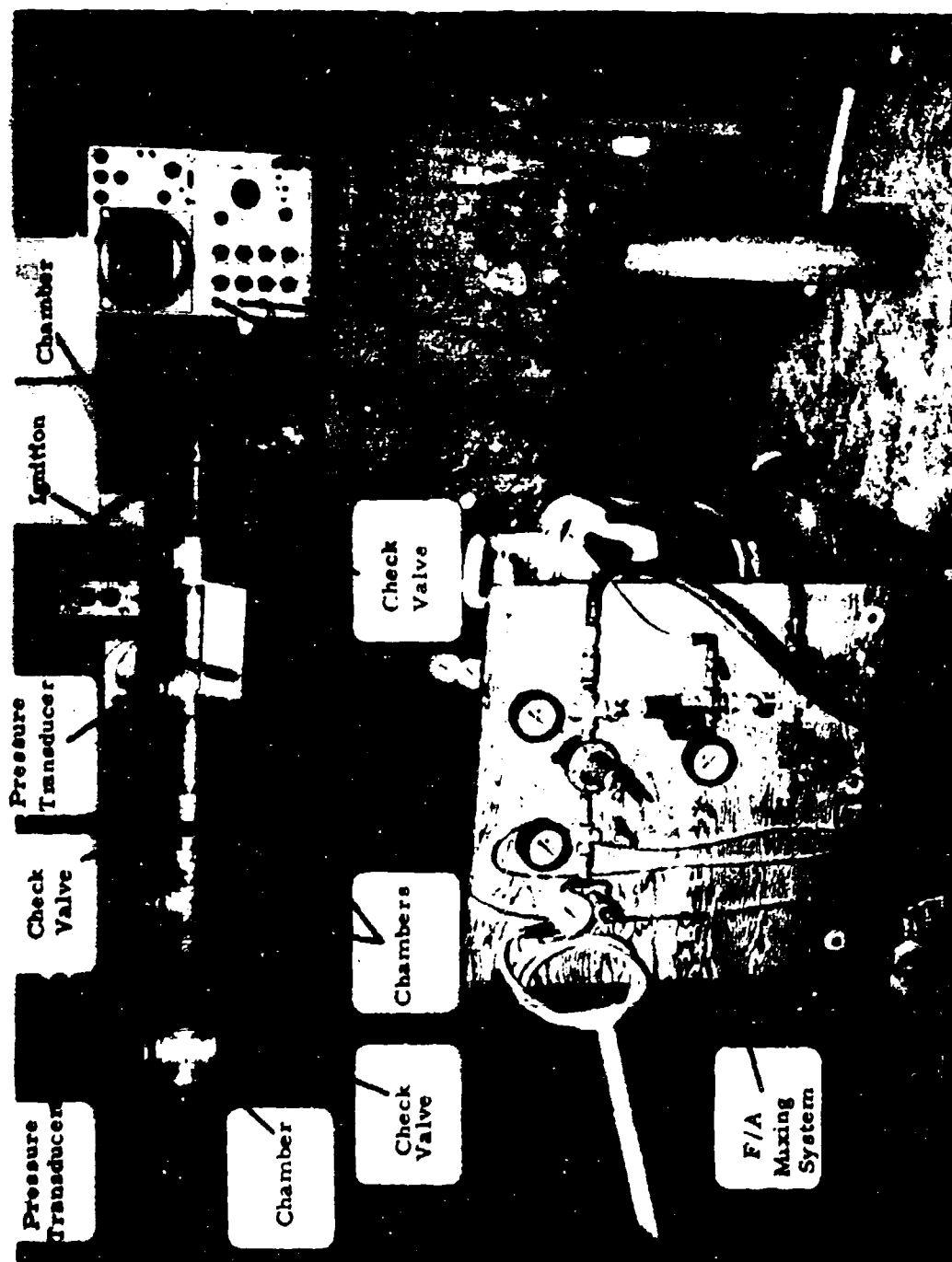
FIGURE 3. MODEL TEST OF THREE STAGE COMBUSTION DEVICE

$\alpha = .12$
1st. Stage Press = 330
2nd. Stage Press = 775 psia
Charge Press = 54 psia
Time - 50 msec/div

FIGURE 4.   MODEL TWO-STAGE COMBUSTION
TEST P-T TRACE WITHOUT CHECK VALVE
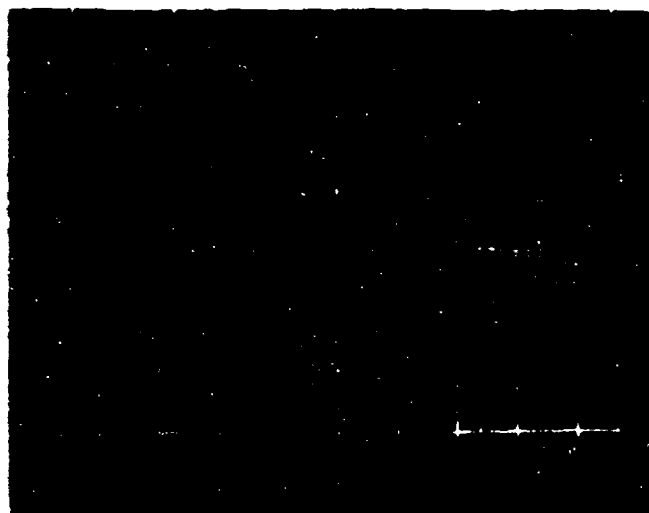
$\alpha = .13$
1st. Stage Press = 240 psig
2nd. Stage Press = 980 psia
Change Press = 54 psig
Time - 50 msec/div

FIGURE 5.   MODEL TWO-STAGE COMBUSTION
TEST P-T TRACE WITH CHECK VALVE

TABLE 1 - TEST RESULTS FOR MODEL STAGE
COMBUSTION WITH ORIFICE

Volume, in.^{3}		Volume Ratio	Charge Pressure	Max. Pressure	Pressure Ratio
V_1	V_2	α	psia	psia	P_b
92.5	12.1	.12	31	351	11.3
92.5	12.1	.12	41	585	14.2
92.5	12.1	.12	54	775	14.3
92.5	32.3	.26	22	279	12.7
92.5	32.3	.26	31	480	15.5
92.5	32.3	.26	41	615	15
92.5	32.3	.26	54	760	14.1
32.3	22.2	.41	22	305	13.8
32.3	22.2	.41	41	505	12.3
32.3	22.2	.41	54	585	10.8
92.5	-0-	1.0	55	320	5.8

TABLE 2 - TEST RESULTS FOR MODEL STAGE
COMBUSTION WITH CHECK VALVE

Volume, in.^{3}			Volume Ratio σ	Charge Pressure psia	Maximum Pressure psia	Pressure Ratio P_h
V_1	V_2	V_3				
Two Stage Combustion with Check Valve						
94.6	14.3		.13	26	450	17.3
94.6	14.3		.13	31	560	18.1
94.6	14.3		.13	54	980	18.1
94.6	34.4		.27	31	476	15.3
94.6	34.4		.27	41	602	14.7
94.6	34.4		.27	54	930	17.2
94.6	94.6		.50	31	361	11.6
94.6	94.6		.50	41	535	13
94.6	94.6		.50	54	585	10.8
Three Stage Combustion with Check Valve						
97	36.7	24.4	.15	26	480	18.5
97	51	55	.27	26	390	15
95	48	71	.33	26	327	12.5

2.   Evaluation

        The limited data acquired with these models demonstrated
that a substantial pressure increase could be obtained by using a low
charge pressure with the stage combustion process.   The results ob-
tained during the testing of the various configurations are shown in
Figure 6.   Several important conclusions can be determined by com-
paring Figure 2, the theoretical, and Figure 6, the actual results.
These results will be individually summarized below:

           a)   The actual process shows a 20% reduction
                in the pressure ratio for a volume ratio
                0.20.   The major contributor to this loss
                would be heat losses; while these losses
                can be expected, it indicated that a 20 psig
                blower would be required in lieu of a 15
                psig blower to obtain the desired 600 psig
                combustion pressure.

           b)   These model tests did not indicate any
                advantage to a three-stage system.   The
                pressure ratio obtained with three stages
                were essentially identical to the two-
                stage configuration.   Examination of the
                three-stage pressure traces indicates
                a very rapid burning rate of the second
                stage with almost a simultaneous burning
                of the third stage.   This rapid burning is
                what caused the lower pressure in the
                final stage.   There essentially was not
                enough time for the unburned gases to
                be compressed before ignition.

           c)   The drop in the pressure ratio for the
                two-stage orifice configuration, at a
                volume ratio of .13 was probably the
                result of too large an orifice for the
                final volume.   Since the final device will
                incorporate interstage check valves,
                no attempt was made to retest with a
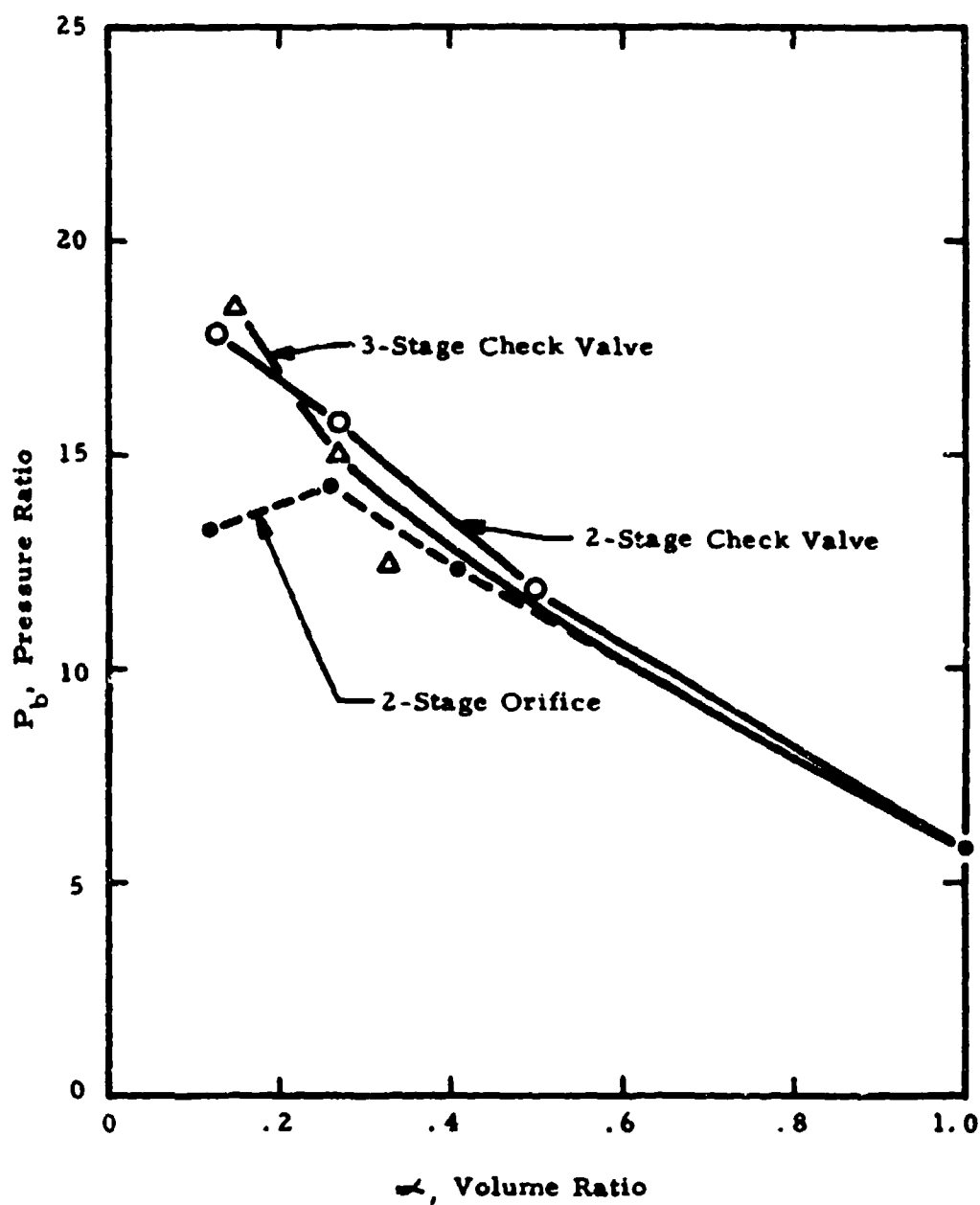   properly sized orifice.

FIGURE 6. RESULTS OF MODEL TEST FOR VARIOUS VOLUME RATIOS

III. DESIGN AND EQUIPMENT DEVELOPMENT

A.   Design Requirements

The basic criteria for the two-stage combustion ripper requires
that all components associated with the combustion device be an integral
part of the tractor. Since the process requires that combustion chambers
be charged with a fuel/air mixture, a blower, fuel pump, ignition system,
and combustion chamber are attached to the ripper section of the tractor.
The entire combustion package has been designed to fit within the dimen-
sional envelope of the tractor and attached to the tractor with a minimum
amount of modification to the existing structure.

Another design requirement for the ripper results from the
primary objective of extending the rock fracturing capabilities of the
D7 tractor to "unrippable" rock. Since the maximum fracturing force
results when only one ripping shank is used, the stage combustion concept
must employ only one shank.

B.   Component Design/Development

1.   Exhaust Valve

The use of a two-stage combustion device requires an
exhaust valve mechanism that will contain the combustible mixture until
peak firing pressure is obtained. This valve must open quickly to allow
the gas to discharge before a sufficient quantity of heat can be absorbed
by the combustion chamber and valve housing. Since the two-stage device
also requires a small final combustion volume, approximately 2 ft.^{3},
the use of the large sliding sleeve valve used on previous REDSOD devices
would be impractical. That particular sleeve valve was an integral part
of the combustion chamber which contained a minimum internal volume
of 2.5 ft.^{3}. It was therefore apparent that a new valve arrangement would
be required for the two-stage system.

Several types of valve designs were considered for this
application, e.g., poppet type, small sleeve valve, and sliding valve.
The poppet valve was the first type of exhaust valve used on the REDSOD
project. Its opening time was rather slow, approximately 30 msec.,
allowing excessive heat losses in the chamber. The sliding sleeve
valve, which was the final valve design used on REDSOD, proved more
reliable than the poppet valve and its opening was approximately 7 msec.
The disadvantage of this sleeve valve is its size for use in this particular
application. In the REDSOD earthmoving device 10 - 15 ft.^{3} of gas had

to be exhausted. For the ripping device only 1.5 - 2 ft.^{3} of high pressure gas is required. The most suitable valve design would, therefore, be one which is compact and quick-opening. The sliding valve was considered to have both these features.

A schematic of the sliding valve is shown in Figure 7. The valve uses a hollow piston which is partially filled with hydraulic oil. The valve is maintained on its seat by constant over-pressure of 40-50 psig. This pressure is applied by a small accumulator mounted on the rear of the tractor. When combustion occurs, the pressure below the piston increases and starts the valve moving upward. As the valve is lifted off its seat, a larger area is exposed to the gases, causing the valve to accelerate. The extension below the valve seat prevents any substantial quantity of gas from escaping down the exhaust duct; allowing additional time for the valve to accelerate. After the valve has traveled 2.5 inches, the discharge area past the valve is equivalent to the exhaust duct area, allowing complete blowdown of the chamber.

In order to stop the upward motion of the valve, a damping wedge is installed above the valve to serve as a hydraulic stop. During operation the air above the valve is compressed and forced into the upper section of the valve housing. The plate located in the upper section is spring-loaded to served as a check valve for the air that is being compressed. When the valve motion is reversed, the check valve closes causing the compressed air to return through a restricting orifice, thereby preventing the valve from being slammed back to the seat. The entire operating sequence for the exhaust valve is shown schematically in Figure 8.

The designs of the stage combustion system and exhaust valve place certain limitations on the size and location of the required components. Since the final combustion stage contains only 2 ft.^{3} of combustion volume, the exhaust duct volume of the ripper shank should be kept to a minimum. This feature is maintained by positioning the exhaust valve directly over the duct-work and using a 4-inch duct to direct the gases to the penetrator tip. The internal volume of the duct-work is approximately 1/2ft.^{3}. The hydraulic snubbing mechanism employed to stop the valve travel requires that the valve travels in a relatively vertical position, to maintain a sufficient quantity of oil in the piston cavity. This feature is usually maintained due to the parallelogram-type linkage of the D7 tractor.

2. Exhaust Duct

The exhaust is ported down the shank by means of 4-inch pipe duct-work and exhausted near the penetrator tip. All gas flow is
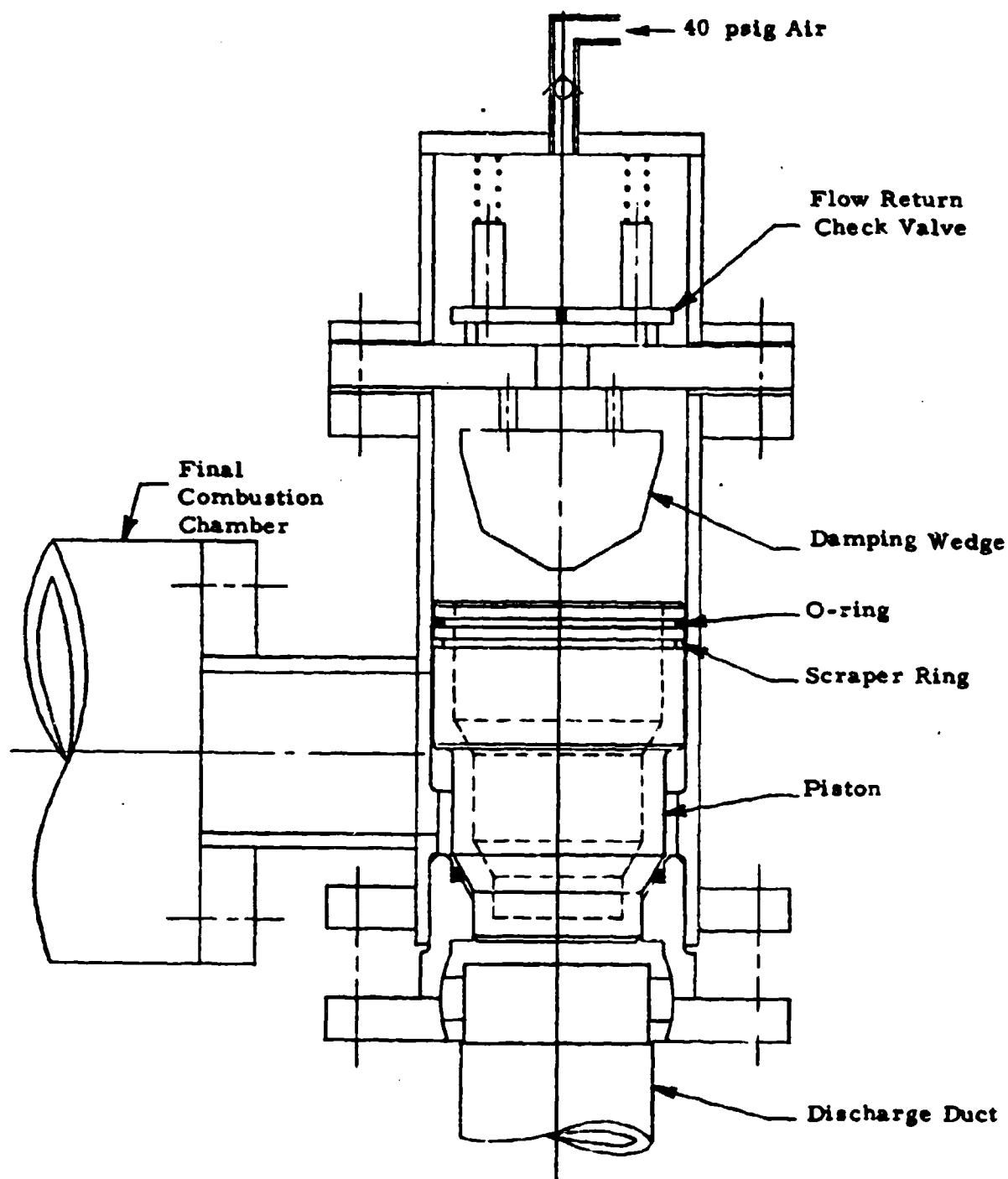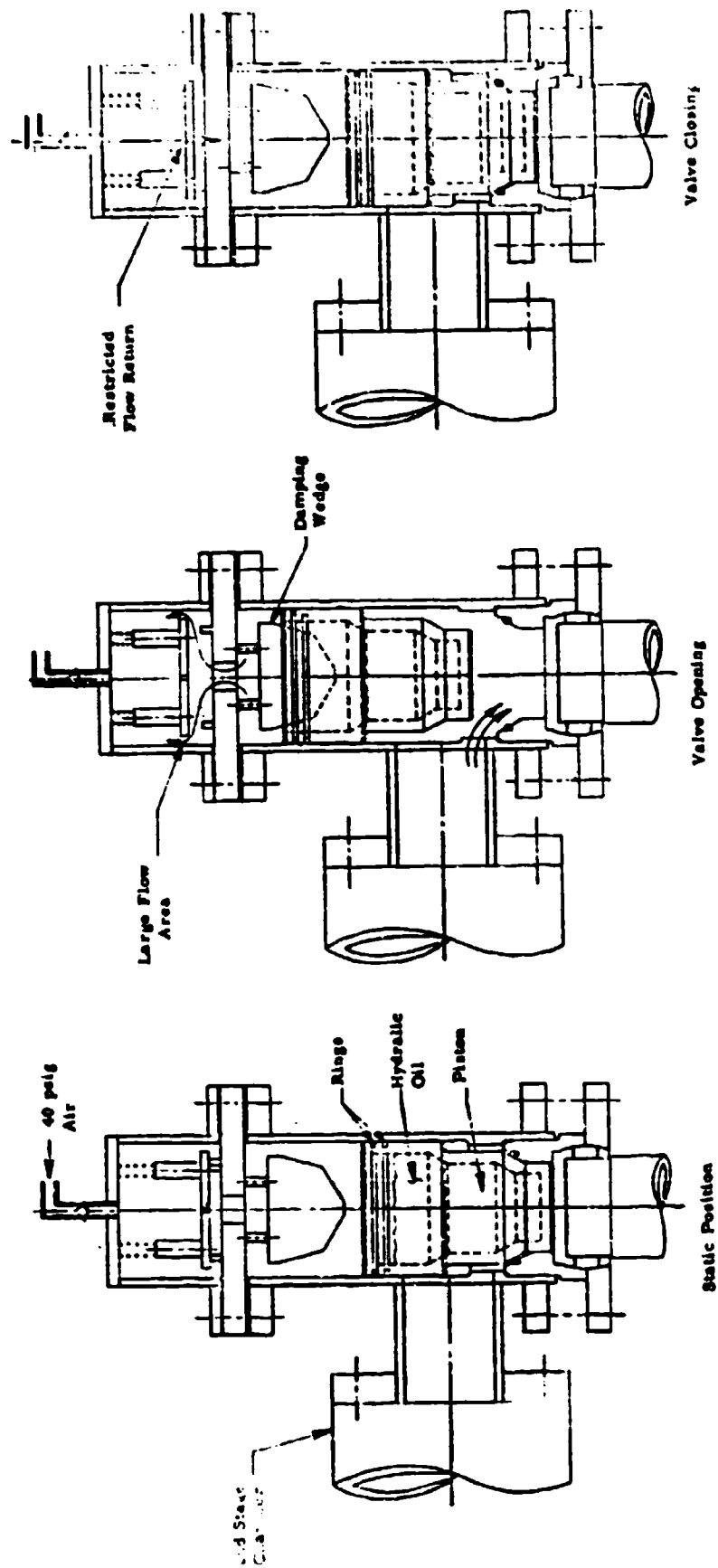
FIGURE 7. SCHEMATIC OF EXHAUST VALVE

FIGURE 8. EXHAUST VALVE OPERATING SEQUENCE

directed below the tip to minimize the problem of duct plugging when being drawn through loosely compacted soil. Previous ripper projects employed a 6-inch pipe duct work, the width of this duct proved to be detrimental in marginally rippable material; the duct, instead of the penetrator tip, was trying to rip the rock. Figures 9 and 10 show the comparison between the standard tooth and the modified tooth respectively.

The evaluation of the modified tooth ripping performance will ultimately be determined during actual ripping tests. During testing, this design may be further optimised for both the standard and explosive ripping conditions it will encounter.

3. Combustion Chamber

The combustion chambers are welded assemblies made from standard 14-inch schedule 40 steel pipe. The larger of first stage chamber is 7.3 ft. in length with a combustion volume of approximately 7 cubic ft. The smaller, second stage chamber is nearly 3 ft. long with a combustion volume of 2 cubic ft. Both chambers have an ultimate bursting strength of greater than 3000 psi; the end plates in the larger chambers are manufactured from 1/2-inch steel plates, while the smaller chamber has 1-inch end plates. Maximum combustion pressure is expected to be approximately 600 psi. In order to prevent combustion back-flow from the small chamber, a specially designed check valve separates the two chambers. This check valve, shown in Figure 11, consists of several small floating poppet type valves. The mass of these valves is minimized to provide a quick response without excessive impact forces on the valve or valve stops. A schematic of the combustion chambers, check valves, nozzles, and igniter locations is illustrated in Figure 12.

C. Control System Design

The initial control system for operating the two-stage combustion device consisted of manually timing the fuel and ignition events. To fire the unit during testing the operator would hold the fuel button on for a certain time period and then press the ignition button. This method allowed a considerable amount of flexibility normally required during the initial testing. For field application, the major events, i.e., fuel, ignition, and chamber purging will be controlled by a preset rotating timing drum. The tractor operator will, therefore, only have one switch, mounted on the ripper control handle, which will activate the rotating time motor. Figure 13 shows the schematic of the entire

FIGURE 9. STANDARD RIPPING TOOTH
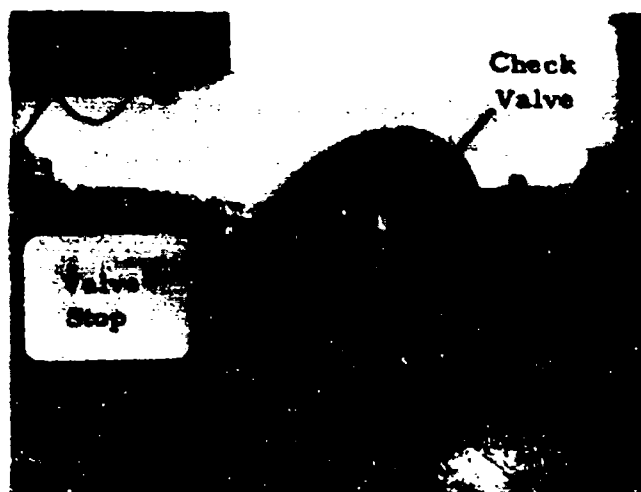

FIGURE 10. MODIFIED RIPPING TOOTH

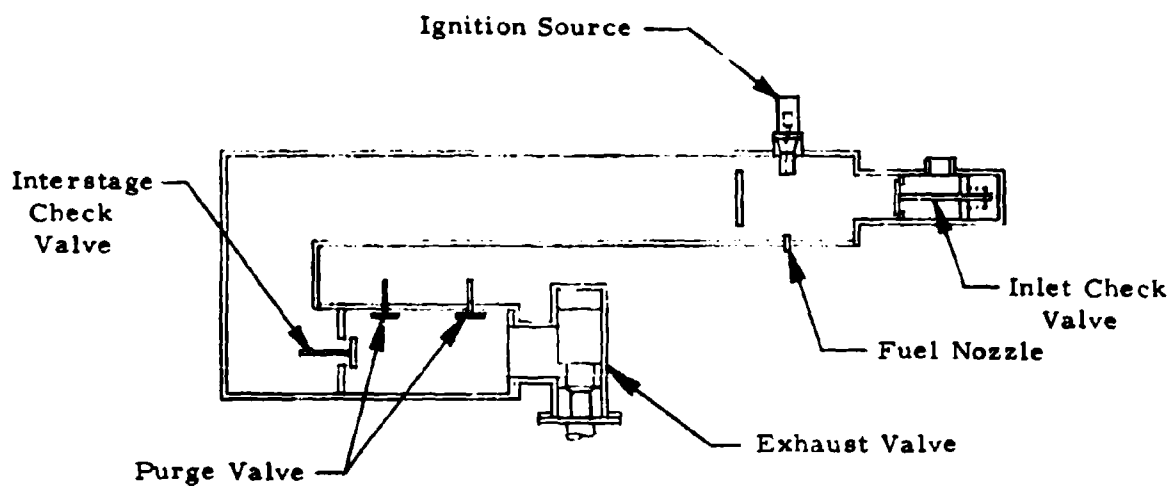FIGURE 11. INTERSTAGE CHECK VALVE

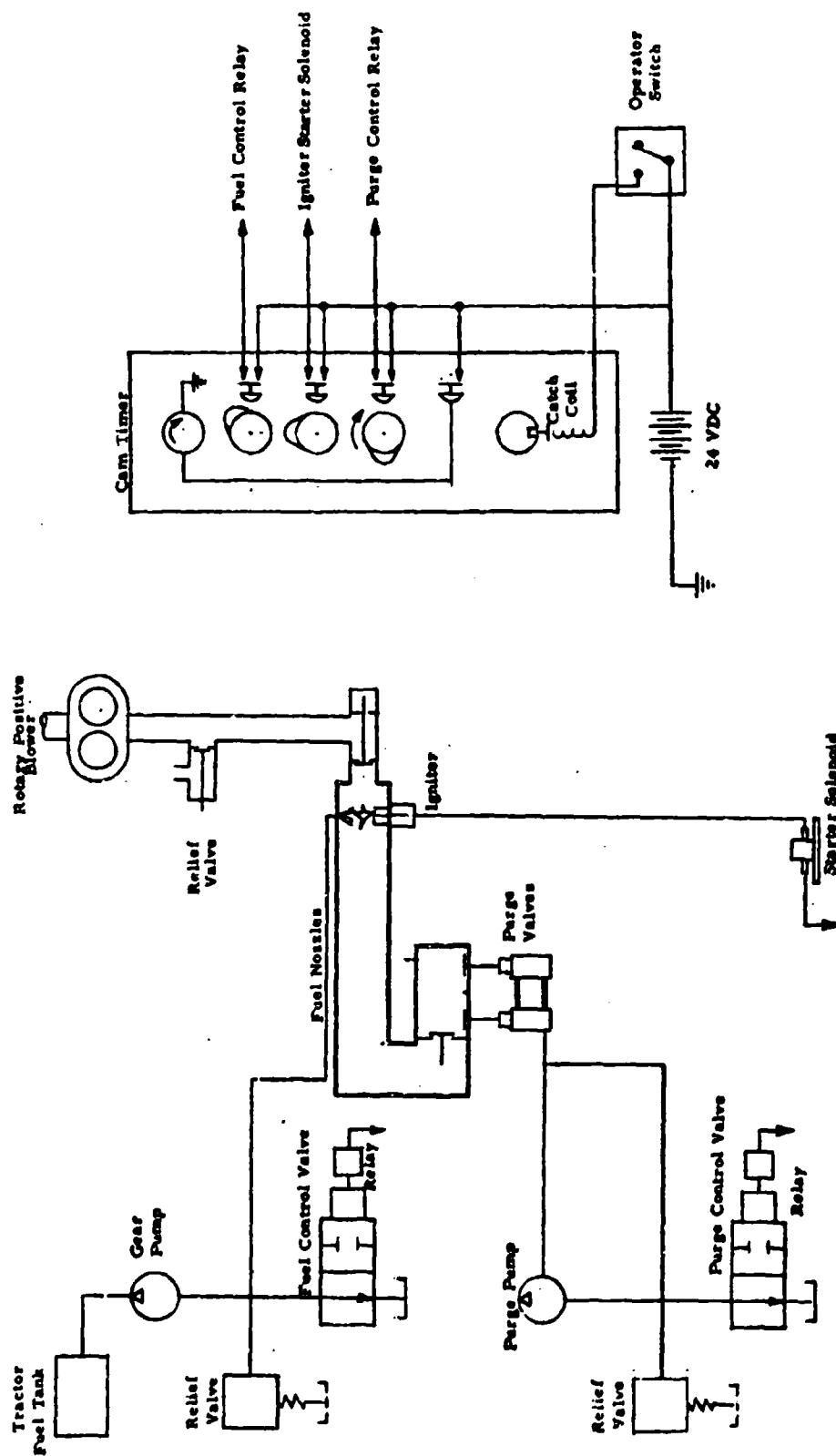FIGURE 12. SCHEMATIC OF COMBUSTION CHAMBER
WITH COMPONENTS

FIGURE 13. SCHEMATIC OF CONTROL SYSTEM

control system. Repetitive operation of the unit is obtained by holding
the "fire button" down. The system may also be operated on a single
shot basis by pushing the fire button once.

The various components used to control the functions
of the fuel or hydraulic valves were manufactured from readily available
commercial equipment. The control valve used for the fuel and hydraulic
operations, shown in Figure 14, is a manually operated valve which has
been modified to be energized by a heavy duty solenoid. This type of
valve is intended for rugged operation and its performance will not be
affected by dust or dirt. Previous REDSOD work indicated an improve-
ment was required in the use of pilot-operated solenoid valves which
would malfunction due to dirt, water or other deposits in the valve
spool. The low tension igniter shown in Figure 15 is a modified auto-
motive start solenoid, which serves as a make-break ignition system.
This type of ignition has proved to be highly reliable in the igniting
of diesel fuel and in continued operation during the testing period.

The roots-type blower used to charge the system can be
driven from the P. T. O. shaft located in the cab section of the tractor.
The blower is in continuous operation, either charging the combustion
chambers or by-passing to the relief valve. Extensive operation of the
tractor in other modes of operation besides ripping would indicate the need
for a clutch to be mounted on the blower.
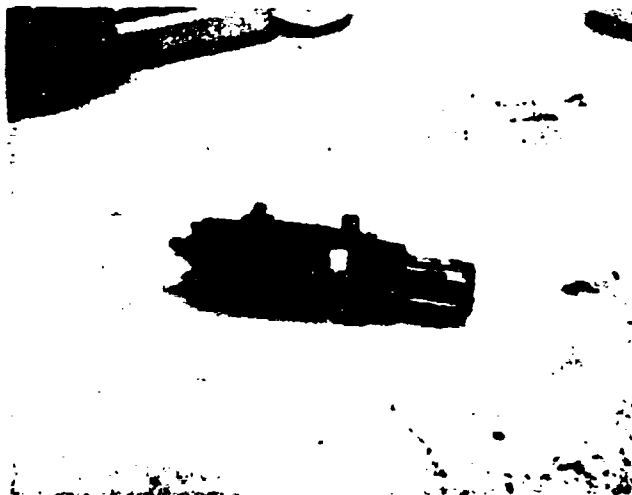
FIGURE 14. FUEL CONTROL VALVE

FIGURE 15. LOW TENSION IGNITER

IV. COMPONENT TESTING

In order to determine the operating characteristic of the various components associated with the two-stage combustion device, a special test fixture was constructed to reduce the noise associated with the rapid discharge of high pressure gas. The fixture shown schematically in Figure 16 consists of several large volume tanks that were used as high pressure air reservoirs and blowdown chambers for the exhaust gases. Utilizing this fixture it became possible to test the components in the vicinity of the laboratory without producing an annoying disturbance to the surrounding personnel.

A.   Exhaust Valve Testing

The exhaust valve was the first component to be tested. This valve is the most critical component of the entire system; if it operates with only a low degree of reliability, the entire advantage of explosive ripping would be reduced.

The exhaust valve operation was initially tested by attaching a 2 ft.^{3} combustion chamber to the valve and discharging the exhaust gases into the blowdown tank. Figure 16 shows the arrangement used for valve testing. The combustion volume of 2 ft.^{3} represents the same volume that would be used on the final two-stage device; however, for testing purposes this chamber was charged to 85 psig with a propane-air mixture. This mixture would produce a peak combustion pressure of 500-600 psi with a rapid pressure rise rate, the same peak pressure and rise rate expected with the two-stage diesel-fueled arrangement.

Pressure transducers were mounted in the combustion chamber and in the upper section of the exhaust valve. The upper pressure trace shown in Figure 17 represents the combustion pressure; the lower trace shows the pressure above the exhaust valve. The charge pressure for this particular "shot" was 60 psig producing a peak pressure of 450 psi in a 10 msec. period. The exhaust valve contained a hold-down pressure of 75 psig; this pressure increases as the valve begins its upward travel, reaching a maximum when the valve is fully opened. Figure 17 shows the pressure trace above the exhaust valve reaching its maximum in a period of 8 msec. This rise rate also indicated that exhaust valve had an opening time of 8 msec. The valve is maintained in the open position by the exhaust gas pressure acting over the lower surface of the valve, and the small air return orifice located in the valve. The valve opening
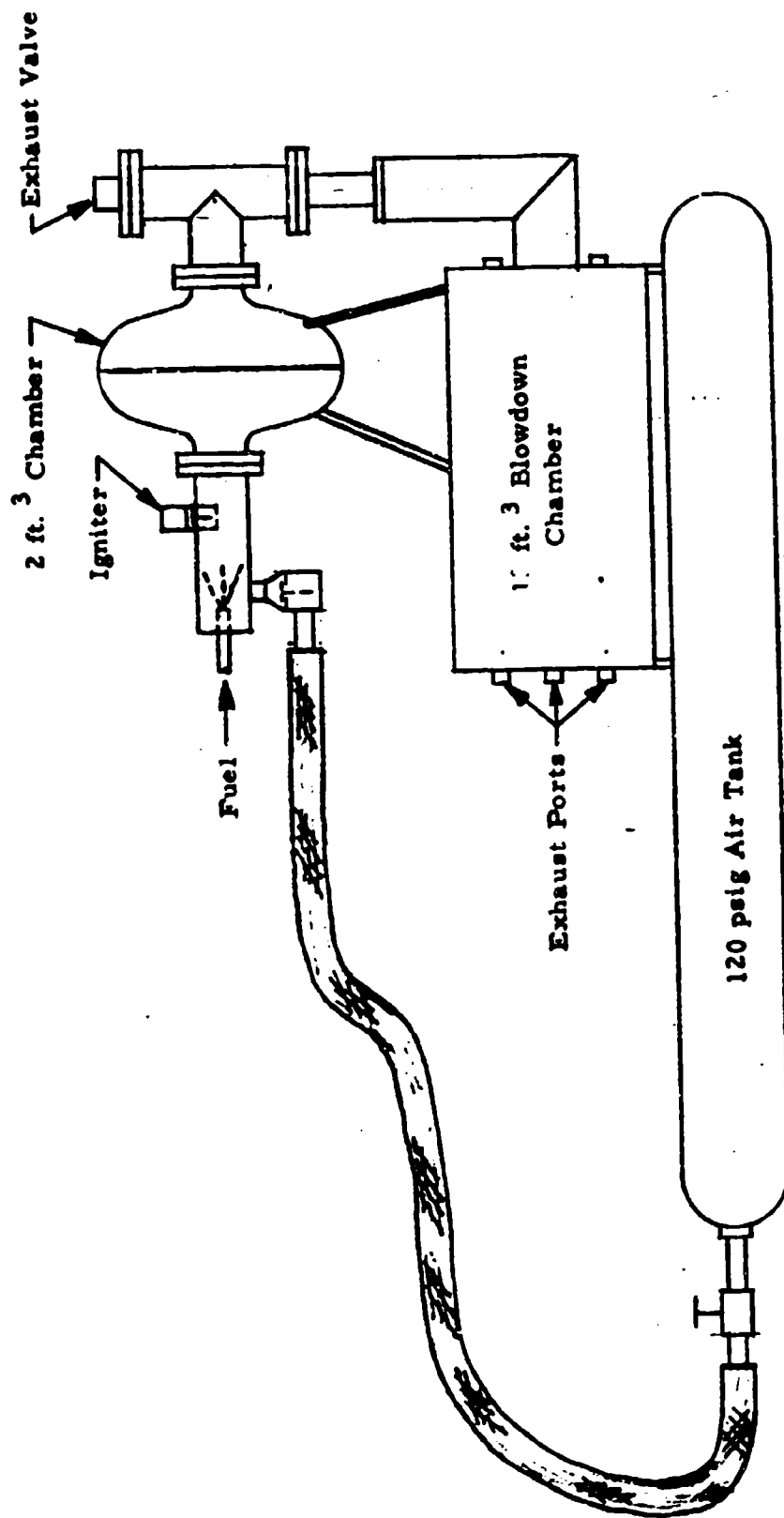
FIGURE 16. SCHEMATIC OF LARGE SCALE TESTING FIXTURE
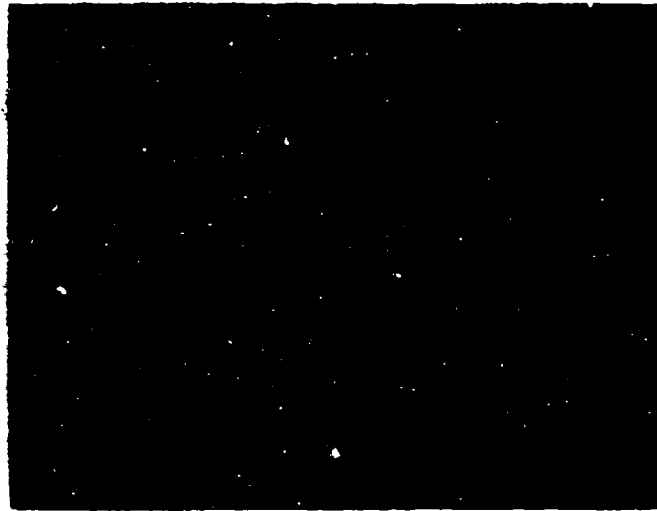
Top Trace, Comb.   Bottom Trace, Exhaust
Chamber   Valve
Charge Press = 60 psig   Valve Press = 75 psig
                           Max. Valve Press =
Peak Press = 450 psig   175 psig

Time - 10 msec/div

FIGURE 17. PRESSURE-TIME TRACE OBTAINED IN COMBUSTION
CHAMBER AND ABOVE EXHAUST VALVE

and closing times is analogous to that of a quick-opening, slow-closing door hinge. Preliminary testing of the valve indicated that quick-opening time is equivalent to that of the previous type of sliding sleeve valves used on REDSOD devices.

Another major concern with the exhaust valve was that of thermal expansion. The valve was fired for 40 shots in approximately a 5-minute period to determine the effect of the hot combustion gases on its operation. Upon completion of this test, the valve was disassembled and inspected. No deterioration was evident in the hydraulic oil or the O-ring used on the valve. The maximum temperature recorded during this test was 275° F on the lower section (near exhaust duct) to 250° F in the center section of the valve body. This temperature corresponds to those obtained on previous REDSOD chambers fired at a faster rate. The upper portion of the valve remained relatively cool during the entire period. The use of hydraulic oil to stop the valve travel and provide a coolant to the valve itself, greatly improves the ability for sustained operation. In the event that over-heating becomes a problem, the hydraulic oil could be recirculated to a coolant tank; however, this problem is not anticipated.

B.   Stage Combustion Testing

The next phase of testing consisted of mounting a two-stage combustion system onto the valve and determining its operation with a 20 psig blower and diesel fuel. Two combustion chambers, slightly smaller in volume than the final chambers, were constructed and attached to the exhaust valve and blowdown tank fixture. Figures 18 and 19 show the size and configuration of the test chambers. These chambers were divided by a floating poppet type check valve which eliminated the backflow of the high pressure combustion gases in the second stage chamber to the lower pressure gases of the first stage chamber. A hydraulically operated purge valve was located downstream of the check valve to purge the large chamber of burnt gases between firings. The igniter and air inlet were both located in the far end of the large chamber with the fuel spray nozzles positioned along the length of both chambers. Pressure transducers were also located in the center section of each chamber for the test period.

Several different nozzles and igniter locations were investigated to determine which position provides the fastest ignition. These locations varied from injecting the fuel into the incoming air stream and placing the igniter downstream of the mixture, to swirling the air and fuel past the igniter located in the combustion chamber. When the proper size nozzle was used to spray the fuel across the igniter, ignition would occur immediately after the ignition button was pushed. This quick ignition
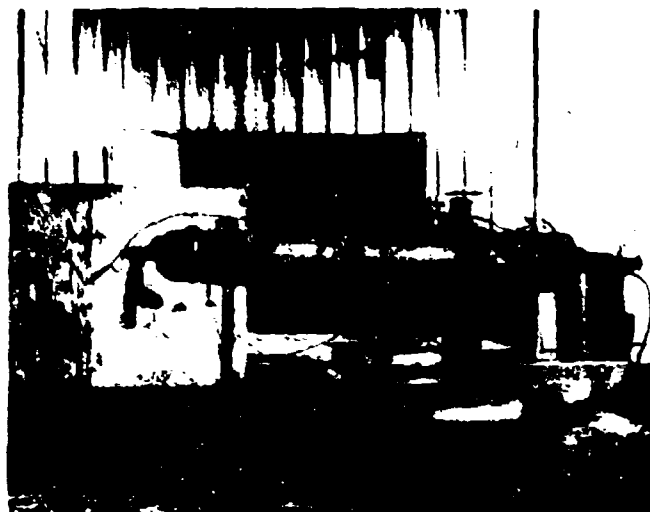
FIGURE 18.   LARGE CHAMBER TEST FIXTURE
(FIRST COMBUSTION STAGE IN FOREGROUND)

FIGURE 19.   LARGE CHAMBER TEST FIXTURE
(SECOND COMBUSTION STAGE AND EXHAUST VALVE IN FOREGROUND)

could be attributed to the low tension ignition system used on the device.

The number and location of fuel spray nozzles in the large, first stage chamber were varied to produce the maximum combustion pressure with the minimum number of nozzles. One disadvantage to this test system was the fact that the nozzles were spraying fuel directly across the combustion chamber. This resulted in a considerable amount of white smoke from the fuel striking the relatively cold combustion chamber walls. The final configuration would have the nozzles spraying along the longitudinal axis of the first stage chamber.

The combustion pressure obtained with this test device reached 500-550 psia in the second stage chamber. This pressure was slightly less than the 600 psig expected. It is believed that this lower pressure was the result of an improper fuel-air distribution. The flame would progress down the chamber reaching pockets of excessively rich or lean fuel-air mixture, burning the fuel droplets only when a sufficient amount of air was available. Regardless of the mixture problem, the pressure obtained is essentially equivalent to that obtained with previous REDSOD devices. Figure 20 shows a typical combustion pressure trace obtained in the second stage combustion chamber. The blower output was 20 psig and the chamber was filled with air before the fuel was injected. A simultaneous fuel-air charging process would, however, produce a more homogeneous distribution. The pressure obtained for this shot was 500 psia. The gradual increase in pressure prior to the rapid rise represents the compression of the second stage from the first stage burning. In this case, the mixture was compressed to 85 psia. The rapid rise results from the flame front entering the second stage through the check valve, causing a strong ignition source, along with the fact that a considerable amount of turbulence is generated as the flow is forced through the check valve.

In order to remove the burnt combustion gases of the first stage, a purge valve was mounted on the second stage combustion chamber. These valves were activated after the exhaust valve had closed, allowing the remaining pressure in the second stage to vent, which in turn caused the 75-85 psi burnt gas pressure in the first stage to open the check valve and vent to the atmosphere. Upon completion of this step, the system would be recycled for another shot.

C.   Full-Size Testing on Tractor

Full-size testing of the staged combustion unit consisted of mounting the same components, e. g. , check valve, purge valve,
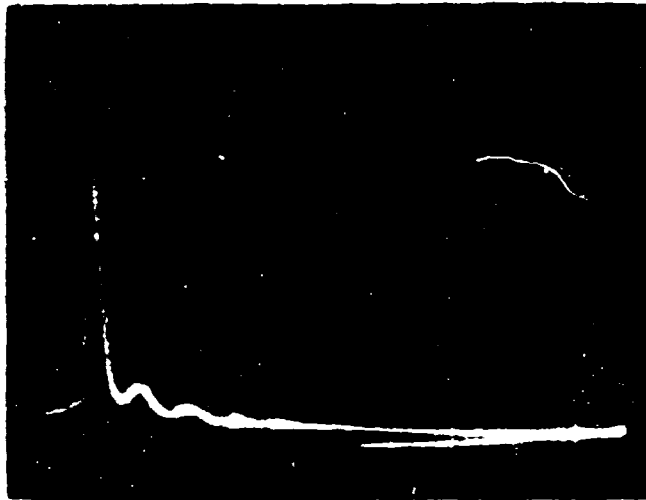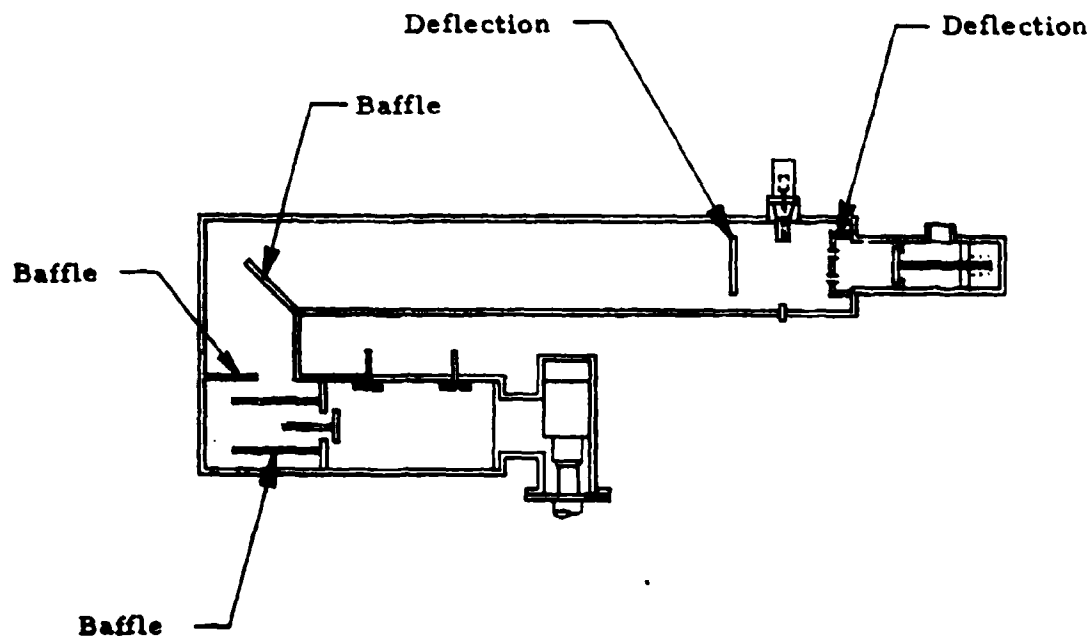
2nd Stage Comb. Trace
Charge Press = 20   psig
Max. Press = 500 psig
Time - 50 msec/div

FIGURE 20.   PRESSURE-TIME TRACE OBTAINED IN
             LARGE SCALE TESTING

exhaust valve, etc., used for the large scale test on the slightly larger final combustion chamber size. The entire combustion system and exhaust duct was then operated on the D7 tractor. Again pressure transducers were mounted in the first and second combustion chambers. Initial testing of the full-size combustion unit indicated a decrease in peak second stage combustion pressure. The large model test configuration produced a pressure of 500 psig while the full-size unit only produced 300 psig. The pressure-time traces obtained for both configurations were examined along with the geometry of the first and second stage chambers. The pressure-time traces indicated a rapid burning of the mixture was occurring in both the first and second stage chambers. This rapid burning produced insufficient time for the unburned gases of the first stage to be forced through the check valve into the second stage. As a result, the flow was choked through the check valve and the volume of unburned gas in the second chamber was considerably less than expected, causing a lower combustion pressure when the mixture ignited. The rapid burning of the mixture was the result of turbulence due to the charging process along with the turbulence that was created as the flow went around the "dog-leg" to the second chamber. This "dog-leg" allowed the flame front to spread out as it reached the end of the first stage, reaching the second chamber before all the gases were burnt in the first stage.

In order to reduce the turbulence and consequently the flame speed, baffle plates and deflectors were installed inside the first stage combustion chamber. These plates were installed near the air inlet to prevent the air from "boring" a hole down the center of the chamber and also at the all section of the first stage to eliminate an increase in the flame front. The final configuration of the first stage combustion chamber and the pressure-time trace obtained with this configuration are shown in Figure 21. Examination of the pressure-time traces shown in Figure 21 indicates that the first stage (bottom trace) is burning in a 0.2 second period. While this time is longer than that of the previous configuration, it is still fast enough to choke the flow through the check valves. The second stage (top trace) shows a peak pressure of 480 psia. This is approximately the same pressure we were obtaining with the large model test unit; however, the geometry and check valve location was different for these two configurations. The rapid pressure rise of the first stage from 60 psig to 140 psig when the second stage ignites may be caused by either of two events. First, the pressure rise could be caused by burning gases that are on the wrong side of the check valve when the second stage ignites. This event is difficult to imagine, considering the number of baffles located in the end section of the first stage. The second possibility is that back-flow is occurring through the check valves before they close. Elimination of either of these problems

SCHEMATIC OF FINAL CONFIGURATION

Charge
  Press = 20 psig
Max. Final
  Press = 480 psig
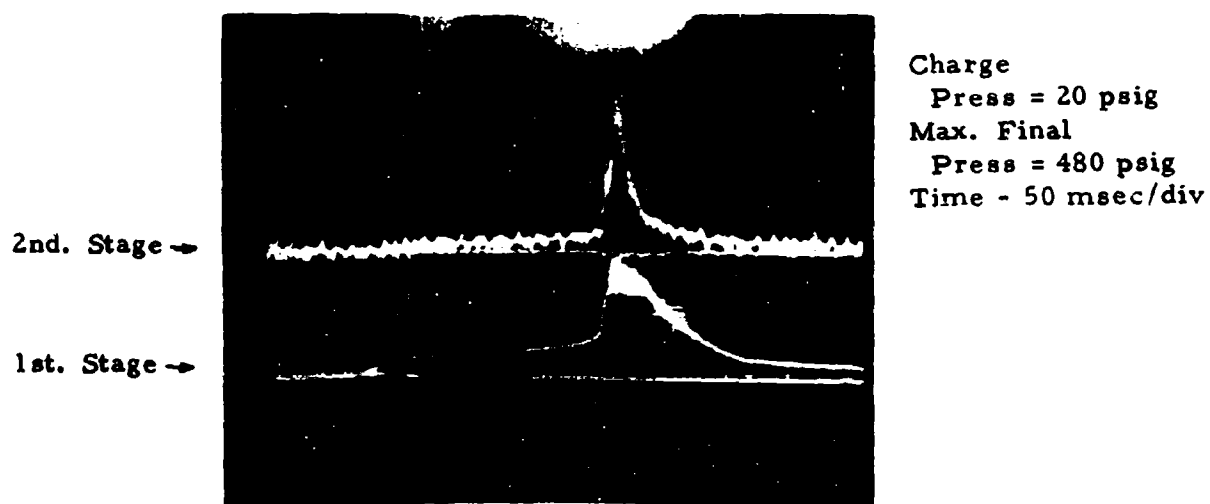Time - 50 msec/div

2nd. Stage →

1st. Stage →

FIGURE 21. PRESSURE-TIME TRACE OF
FINAL CONFIGURATION

would result in a higher second stage pressure. The exhaust valve hold-down pressure during this series of tests was 60 psig; the combustion pressure required to start the valve in motion is approximately 240 psia with a valve opening time of 0.007 seconds. This would indicate the possibility of premature valve opening. Increasing the valve hold-down pressure to 100 psig would allow more time for pressure build-up in final stage.

While the full combustion pressure was not obtained during this test period, a better understanding of the events and mechanisms involved with the two-stage combustion unit has been obtained. By incorporating the previously mentioned changes of eliminating turbulence, slowing down the burning rate, reducing back-flow and increasing hold-down pressure, the unit can be expected to produce a higher peak combustion pressure. The changes that have been utilized, namely installing deflectors and baffler, increased the pressure from 300 psi to 500 psi; this was essentially only one change which produced a substantial effect.

During the automatic firing mode, it became apparent that the capacity of the blower, 350 scfm, would require 3 seconds to charge the combustion chambers to 20 psig. This charging time would limit the firing rate of the unit to 15 shots per minute, allowing 1 second for exhausting and purging the chambers. This firing rate may be increased by positioning two blowers in tandem operation. The timing pressure of fueling, ignition, and purging the system for the automatic firing mode is illustrated in Figure 22.
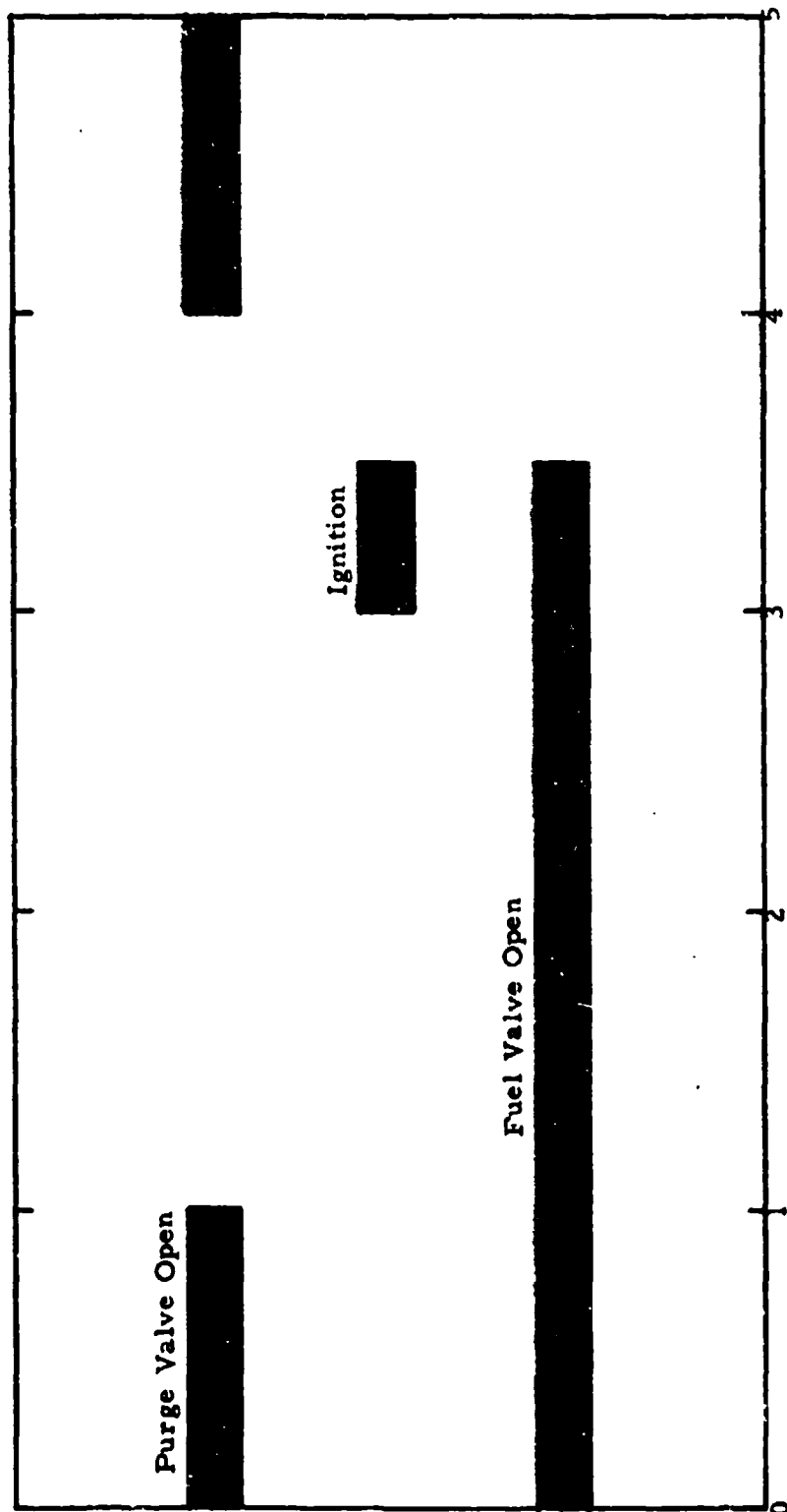
FIGURE 22 TIMING SEQUENCE OF TWO-STAGE COMBUSTION

V. SYSTEM MALFUNCTIONS

The overall operation of the two-stage combustion system could only be determined after a sufficient period of actual field testing. This report is concerned only with the development of the concept without actually field testing the unit. During the course of this development, several systems and components were designed and tested; the problems encountered with these components may provide an insight to the expected field reliability of the unit.

A.   Combustion Related Malfunctions

The only malfunctions that have occurred during testing of the large combustion chambers may be attributed to an improper fuel mixture causing excessive carbon build-up on the component. This problem was first noticed on the low-tension igniter. The original igniter configuration consisted of the center electrode sliding in an insulated phenolic bushing. The clearance hole in this bushing would become plugged with carbon due to gas leakage past it, eventually causing the system to short out or preventing the required movement of the electrode. This initial configuration was modified to incorporate the phenolic bushing inside a steel bushing. In this manner, the leakage was directed between two steel components eliminating the carbon problem previously mentioned.

B.   Structural Failures

During the development test period only two minor structural type failures occurred; this happened to the interstage check valve and an exhaust valve gasket. The initial design of this check valve consisted of a 6" diameter, 5/8" thick steel poppet type valve. This valve was required to prevent the back-flow of combustion gas from the second stage chamber to the first stage chamber. The valve was maintained in a floating position allowing it to open to the second stage when initial burning occurred and rapidly closing when the second stage ignited, having a lift of approximately 1/4 inch. After several test firings, the jam nuts restricting the valve lift became loose, allowing the lift to increase on successive shots. The acceleration of this valve and its weight, approximately six pounds, finally caused the stem to fail in tension. The valve was then blown into the second stage chamber, reducing the effect of two-stage combustion. To eliminate

this problem, a new interstage valve was designed which utilizes seven smaller and lighter valves. The lift of these valves is limited by a stop, welded to one side of the supporting structure. The concept has continued to produce satisfactory results for the duration of the project. One significant advantage to this multi-valve concept is that if one or two valves should fail to operate, the resulting flow area is small enough to provide sufficient pressure build-up in the second stage before excessive back-flow occurs.

The second failure that occurred was a blown gasket, located in the exhaust valve housing. The crush area for this gasket was increased and the gasket continued to function for the duration of the project.

Both of these failures and problem areas associated with the combustion process can be considered relatively minor compared to the overall scope of the project. Several additional problem areas will probably be exposed during the actual field testing; however, the simplicity of the entire system and operation of the various components to date indicate a high probability of satisfactory operation.

VI. DISCUSSION OF RESULTS

In this section, the results described earlier for the various phases of the project will be categorized and discussed.

A. Explosive Ripper Design

1. Tractor Attachment and Modifications

The overall configuration and minimum amount of auxiliary hardware associated with the two-stage combustion indicates that the entire system may be furnished as a bolt-on type kit. Figures 23 through 25 illustrate the area of the D7F tractor where the two-stage combustion unit is mounted.

The combustion chambers are bolted to the rear linkage of ripper attachment while the exhaust valve is bolted to a stud plate which is welded to the tractor. Slight modification to one ripper tooth attaching bracket and one of the push blocks were required to mount the system on the tractor. Future chambers and exhaust valves could easily be modified to eliminate any tractor modification with the exception of bolt holes, necessary for attachment points. The attachment points for the first combustion stage is shown in Figure 26.

Figure 27 shows the extension shaft which could be used to belt drive the blower, fuel pump, and hydraulic pump component. In order to reduce the time required to mount this extension shaft, a hydraulic pump could be attached to this power take-off shaft, which in turn would operate a hydraulic motor to power these components. The only tie-in point to the tractor system is at the fuel tank and battery box.

The entire two-stage combustion system, with a modified ripper tooth containing the exhaust duct could be installed on the tractor in approximately 8 hours, requiring two people. The weight of the entire system, approximately 1,000 lbs., would require the use of a standard Army crane. The system could be left on the tractor during normal ripping operation and only be activated when required.

The control functions used on the two-stage combustion system require a minimum amount of understanding on the part of the operator. These controls require the operator to determine if repetitive
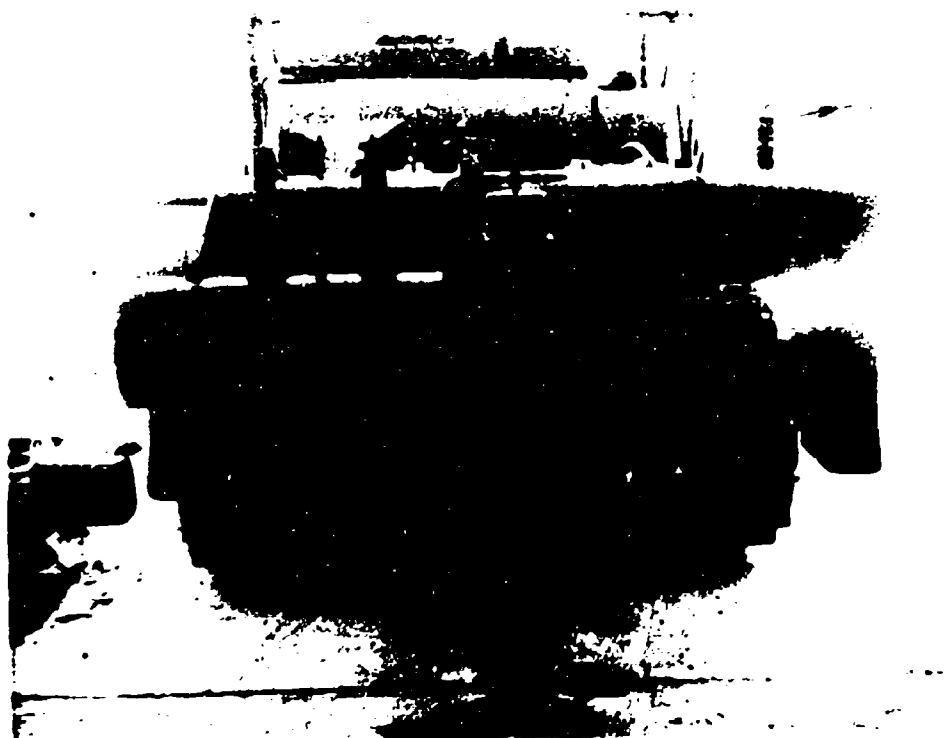
BEFORE

AFTER

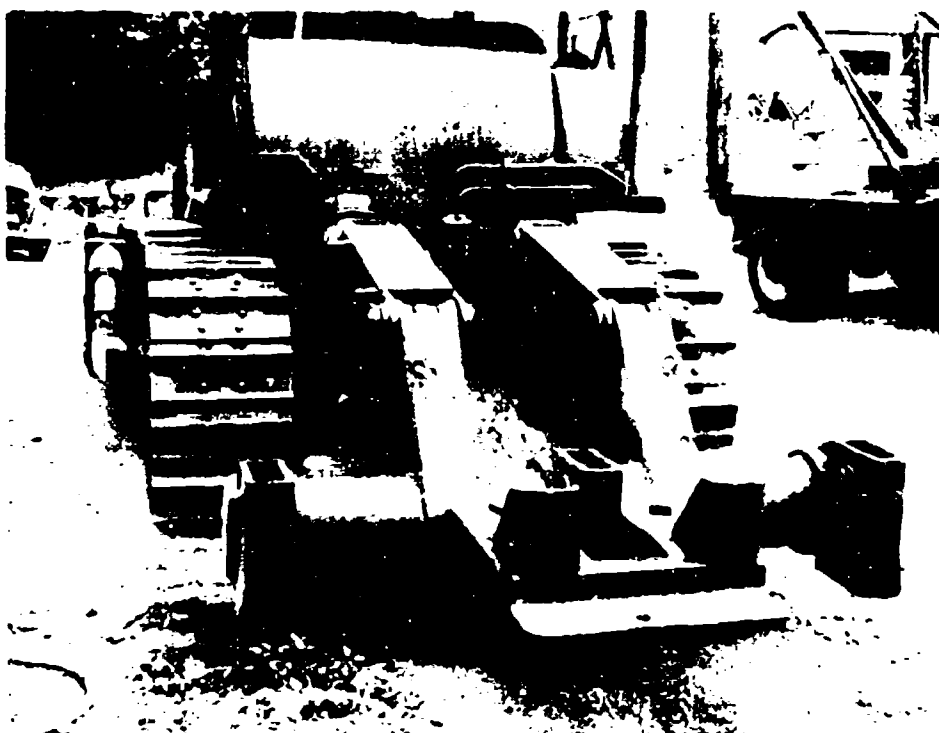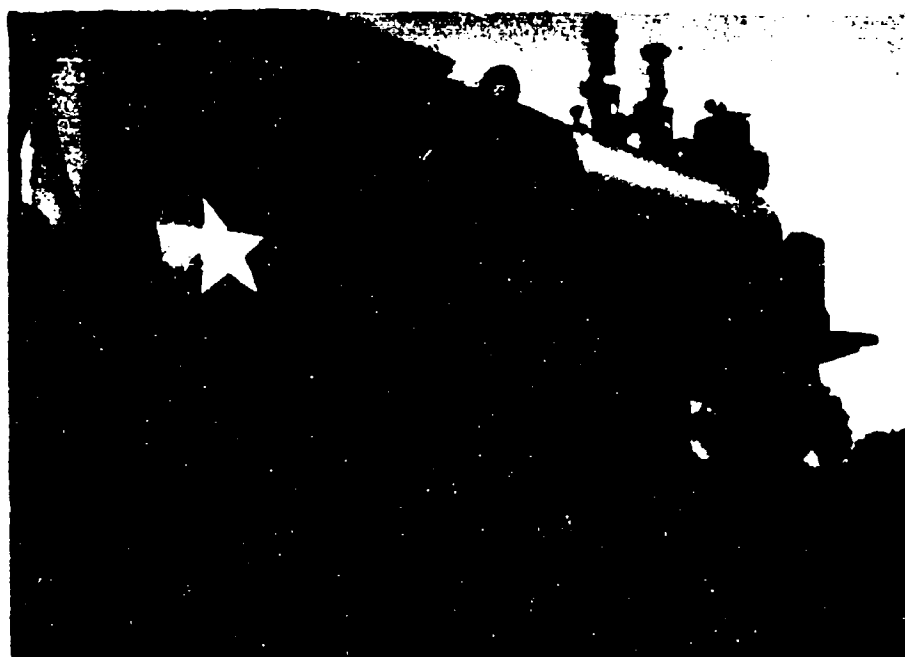FIGURE 23. 3/4 VIEW OF D7F TRACTOR

BEFORE

AFTER

FIGURE 24.   REAR VIEW OF D7F TRACTOR

BEFORE

AFTER
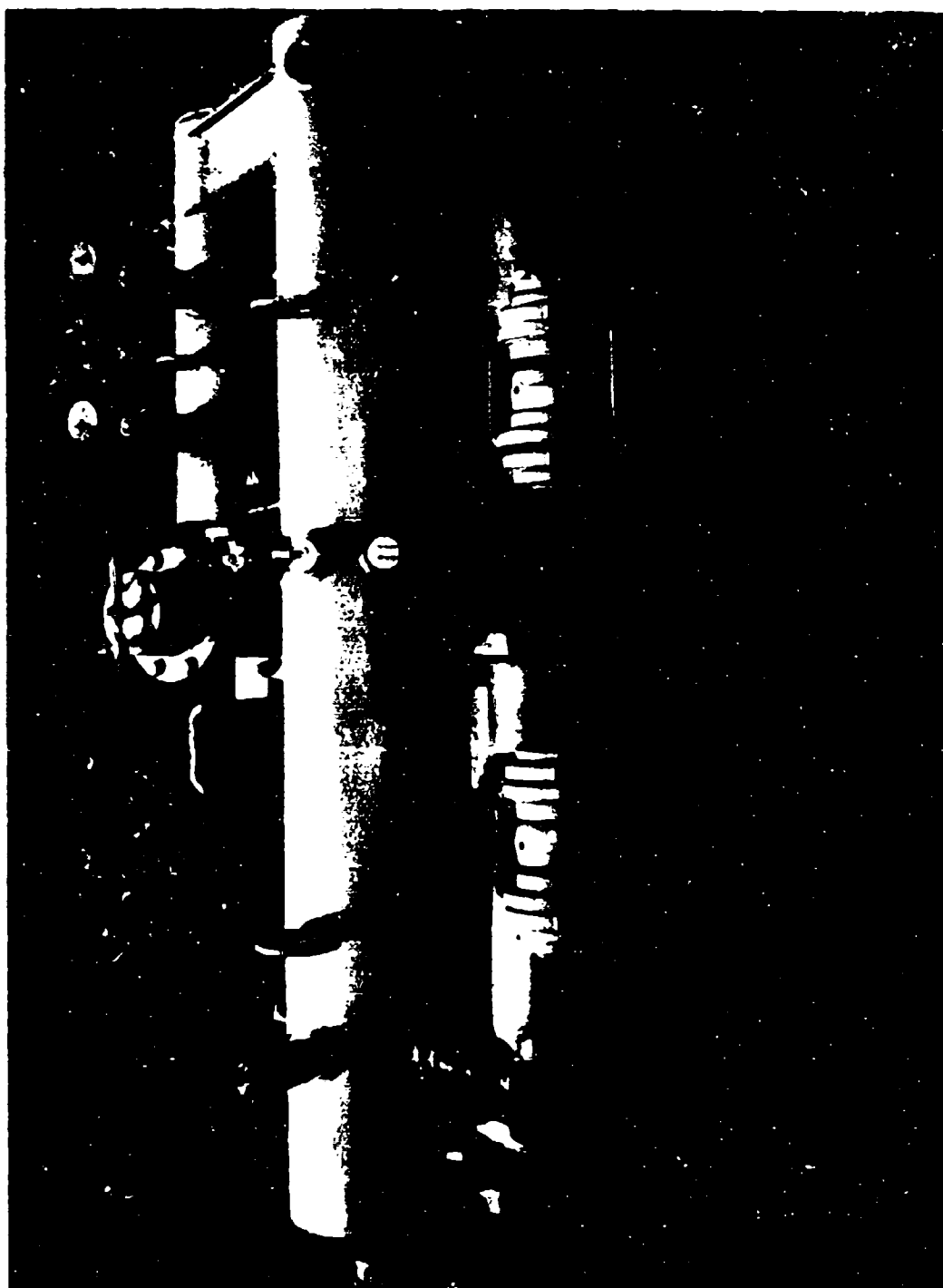
FIGURE 25.. 3/4 VIEW OF D7F TRACTOR

FIGURE 26 - FIRST STAGE ATTACHMENT POINTS

FIGURE 27. P. T. O. SHAFT USED TO DRIVE BLOWER

or single shot application is required. Upon initiation of either of these operating modes, the remainder of the events, fueling, ignition, etc., occur automatically. The control system involves no special electrical or mechanical devices but rather a single switch-operated 24 VDC system.

B.   Two-Stage Combustion Energy

The diesel-fueled, spark ignited, two-stage combustion device produced the same energy release as that obtained on the previous repetitive explosive device developed for rock ripping. The change in fuel from propane to diesel, gasoline, natural gas, or any other type of hydrocarbon fuel would not appreciably change the energy release obtained. The main advantage to the two-stage combustion system is the fact that a large auxiliary air compressor can be replaced by a small tractor mounted blower and no special fuel is required. Continual staging of the combustion chamber is a possible means of obtaining even higher combustion pressure; however, the temperatures and pressures that could occur in a third or fourth stage may be sufficient to compression ignite the diesel fuel mixture before it reaches maximum staging compression pressure.

C.   Predicted Performance

It should be stated that no field testing of the two-stage combustion device was performed; however, the predicted performance of the device may be obtained by comparing it to the previously tested explosive rock ripper. Previous ripping tests were performed with both a 5 cubic foot and a 2.5 cubic foot combustion chamber. The fracture volume and width of an explosive ripping pass produced by either of these chambers was not distinguishable. The present two-stage device has a final combustion volume of 2 cubic feet. The final peak combustion pressure obtained in both configurations is approximately 600 psi. The maximum firing rate for both devices is also identical: approximately 10 shots per minute. Based on the fact that the end components of both ripping devices are essentially identical, it appears logical that the end result would be a same, i.e., nearly a two-fold increase in production rate over that of a standard ripper in marginally rippable rock.

D.   Operating Requirements

1.   Fuel
The two-stage combustion device is designed to operate using

the same diesel fuel tank as the tractor. Since the combustion event is initiated by a spark, the fuel mixture used is near stoichiometric requiring a considerable amount of fuel for the 9 cubic feet of combustion volume. The quantity of fuel used with this device will depend upon its firing rate. Figure 28 illustrates the fuel consumption of the device as a function of firing rate and fuel-air ratios.

2. Electrical

The electrical power for the igniters and control box is supplied by the tractors 24VDC power supply. The amount of power drawn off the batteries is minimal and would not be a drain on the tractor's charging system.
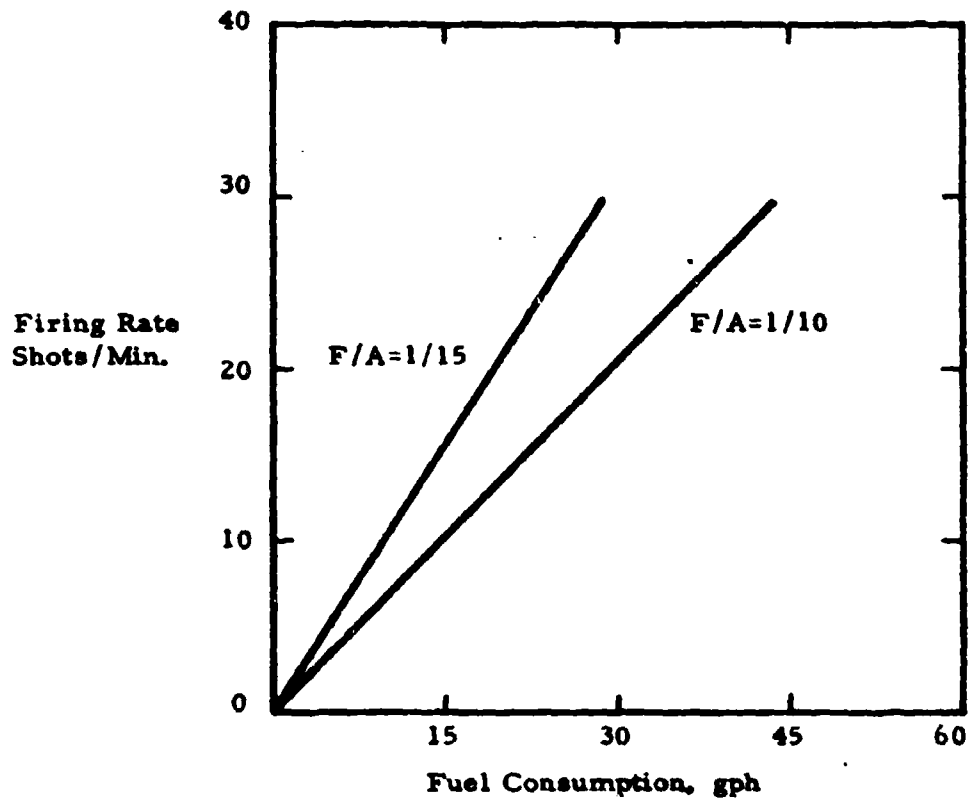
FIGURE 28.   FIRING RATE VS. FUEL CONSUMPTION

VII. SAFETY

The operating method of the explosive ripper requires that a certain amount of safety consciousness be exercised by the vehicle operator. This safety consciousness should be extended to both the operator and surrounding area where the ripper is employed.

The noise produced by the sudden release of high pressure combustion gases would be considered objectional to the tractor operator. In addition to the noise generated by the combustion process, a considerable amount of noise will result from the use of the blower mounted near the cab region of the tractor. Standard ear muff type protection is recommended for the operator while the device is being employed for explosive ripping.

A certain amount of rock throw (approximately 30 yards) can be expected, largely directed to the rear of the tractor. A deflector shield attached to the lower portion of the ripper linkage would prevent an upward rock trajectory. However, operator protection, such as a screen cab is required, since occasional fragments fail to follow a general trend.

During operation of the unit in either the automatic or manual firing mode, the combustion chambers and exhaust duct may reach a temperature of 300° F. Appropriate warning sign indicating that the chambers are hot should be posted on the tractor.

VIII. CONCLUSIONS AND RECOMMENDATIONS

From the test data presented in this report, it is concluded that:

1. The large compressor previously required with REDSOD can be replaced by a small tractor mounted blower employing the staged combustion principle.

2. The peak combustion pressure obtained with a two-stage combustion unit is equivalent to the combustion pressures previously obtained on REDSOD.

3. The entire combustion system can operate utilizing the on-board diesel fuel and electrical system of the D7 tractor.

4. The entire stage combustion system can be furnished as a kit, costing less than 10% of the tractor and being easily adapted to a standard D7F tractor with a ripper attachment.

From the results of this contract, it is recommended that:

1. Field testing be performed on the two-stage combustion device to determine its reliability and define its performance limits.

2. Additional model and/or large scale testing be performed to obtain higher combustion pressure using three or possibly four stages of combustion. The exact combustion pressure required to fracture the various rock formation is not known; however, increased pressure should provide a corresponding increase in production.

APPENDIX A

Analysis of Flame Motion
in Enclosed Spaces

The analysis of the flame motion in the combustion chambers will be investigated for the following cases:

1. No back flow, which assumes no recompression of the burned gases.

2. Back flow situation, condition when recompression of the burned gases can occur in one chamber.

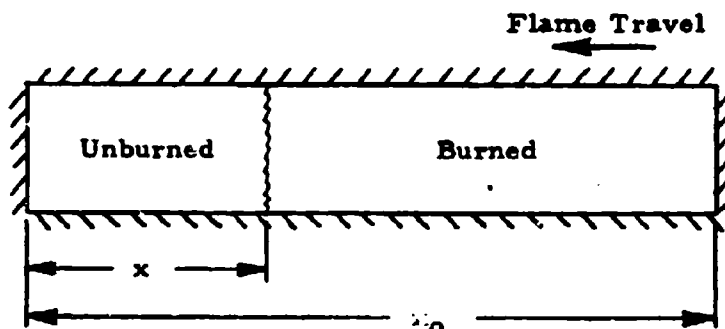The system to be analyzed is shown schematically in Figure 1-A:

FIGURE 1-A

All properties of the system (p, T, ρ, M) will be considered relative to these initial conditions. The method of computation utilized is to compute the change in density in the unburned gas for each increment of flame travel. Each increment of burning will occur at a constant pressure. The combustion assumption is that a perfect gas is involved and that the combustion produces a pressure rise which is constant regardless of density or temperature. Temperature before burning will occur adiabatically.

By the definition of density we have:

$$\rho = \frac{M}{x} \tag{1}$$

Expanding this to account for the burning of an incremental mass and the flame advancing an incremental distance.

$$\Delta\rho = \frac{M}{x}\left[\frac{1 + \Delta M/M}{1 + \Delta x/x} - 1\right] = \rho\left[\frac{\Delta M}{M} - \frac{\Delta x}{x}\right] \tag{2}$$

This is still a general equation which does not account for the no back flow condition. The following equation can be used to relate ΔM and Δx, which also includes the assumption about back flow. Consider the element shown in Figure 2-A:
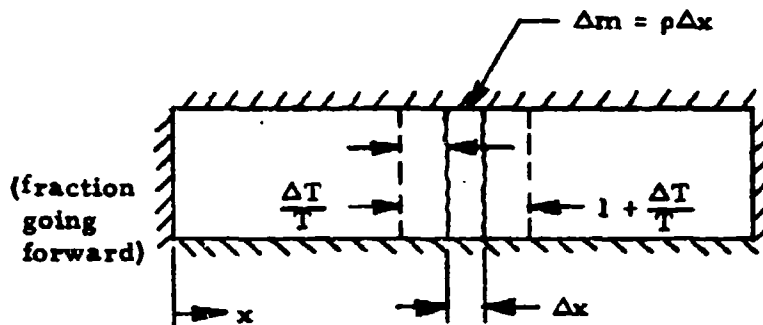

FIGURE 2-A

Since the burning is assumed to occur at constant pressure, the original width of the element, Δx, is increased by the volume ratio:

$$\frac{\text{New Volume}}{\text{Old Volume}} = \frac{T + \Delta T}{T} = 1 + \frac{\Delta T}{T}$$

Therefore, the new element of width becomes:

$$\Delta x = \frac{\Delta m}{\rho}\left[1 + \frac{\Delta T}{T}\right]$$

The term $\Delta T/T$, the volume increment can now be modified for the case of back flow or no back flow. The above equation now becomes

$$\Delta x = \frac{\Delta m}{\rho}\left[1 + \frac{\Delta T}{T}(BFF)\right] \tag{3}$$

where BFF represents the back flow function. In the case of no check valves, the volume is divided in proportion to the length ratio; therefore, BFF is equal to x. In the case of an infinite number of check valves the expansion moves forward and BFF equals 1.0.

Combining equations (1) and (3)

$$\frac{\Delta x}{x} = \frac{\Delta \rho}{\rho} = \frac{\Delta M}{M}\left[1 + \frac{\Delta T}{T}(BFF)\right] \tag{4}$$

Combining equations (2) and (4) we can obtain either density as a function of mass or position. Therefore, to obtain $\rho = \rho(M)$

$$\frac{\Delta\rho}{\rho} = -\frac{\Delta M}{M}\frac{\Delta T}{T} \quad (BFF)$$

(5)

For the use of no back flow BFF = 1.0 and adiabatic compression, equation (5) becomes

$$\left(\frac{\Delta\rho}{\rho}\right)\rho^{k-1} = -\frac{\Delta M}{M}\Delta T$$

(6)

This equation may now be integrated as

$$\frac{1}{\Delta T}\int_{x_0}^{x}\rho^{k-2}\,d\rho = -\int_{x_0}^{x}\frac{dM}{M}$$

(7)

The limits of integration, remembering that all terms are normalized, become: $M(x_0) = 1$, $M(x) = M$ at $x = x_0$, $\rho^{k-1} = 1$.

Equation (7) becomes

$$M = \frac{1}{e^{\frac{\rho^{k-1}-1}{\Delta T(k-1)}}}$$

or by substituting Equation (1)

$$x = \frac{1}{\rho e^{\frac{\rho^{k-1}-1}{\Delta T(k-1)}}}$$

(8)

In a similar manner, using equation (5) for the case of back flow, BFF = x, we obtain

$$M = 1 + \frac{1}{k\Delta T} - \frac{\rho^k}{k\Delta T} = 1 - \frac{1}{k\Delta T}(\rho^k - 1)$$

(9)

or by substituting Equation (1)

$$x = \frac{1}{\rho}\left(1 + \frac{1}{k\Delta T}\right) - \frac{\rho^{k-1}}{k\Delta T}$$

(10)

The term ρ^k is equivalent to the pressure ratio p, therefore, for the case of back flow occuring in one combustion chamber, the standard REDSOD practice eq. (10) defines the value of ΔT for a pressure ratio of 6.

for $M = 0$ $\qquad k\Delta T = p - 1$

or $k\Delta T = 5$ $\qquad \Delta T = 3$

Since x is the volume fraction of the chamber in question, it can be defined as α and the pressure ratios for constant volume burning become:

$$\frac{p}{p_c} = \frac{T + \Delta T}{T} = 1 + \frac{\Delta T}{\rho^{k-1}} \tag{11}$$

The charge pressure p_c is also equal to ρ^k, therefore, equation (11) relative to the initial conditions, and at constant volume, becomes

$$p_b = \rho^k \left(1 + \frac{k\Delta T}{\rho^{k-1}} \right) \tag{12}$$

Substituting the volume ratio α for x, equation (10) for the case of back flow becomes

$$\alpha = \frac{1}{\rho} \left(1 + \frac{1}{k\Delta T} \right) - \frac{\rho^{k-1}}{k\Delta T} \tag{13}$$

Likewise for the no back flow condition equation (8) becomes

$$\alpha = \frac{1}{\dfrac{\rho^{k-1} - 1}{\rho e^{\Delta T(k-1)}}} \tag{14}$$

Equations (12), (13), and (14) were used to obtain the theoretical curves shown in Figure 2.